

Technical Note

Design consideration of low temperature differential double-acting Stirling engine for solar application

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Abstract

This paper presents design considerations to be taken in designing of a low temperature differential double-acting Stirling engine for solar application. The target power source will be a thermosiphon solar water heater with integrated storage system, which will supply a constant source temperature of 70 °C. Hence, the system design is based on a temperature difference of 50 °C, assuming that the sink is kept at 20 °C. During the preliminary design stage, the critical parameters of the engine design are determined according to the Schmidt analysis, while the third order analysis was used during the design optimisation stage in order to establish a complete analytical model for the engine. The heat exchangers are designed to be of high effectiveness and low pressure-drop, and are made from a 0.015 m tube, while the porosity of the steel wool of 0.722 is used for the regenerator matrix. Upon optimisation, the optimal engine speed is 120 rpm with the swept volume of 2.3 l, and thus the critical engine parameters are found to be the bore diameter of 0.20 m. In addition, the volumes of heater, cooler and regenerator are 1.3 l, 1.3 l and 2.0 l volumes, respectively.

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1. Introduction

In the new millennium, the human race is facing the ultimate challenge of how to achieve the goal of a cleaner and healthier environment for the generation to come.

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Nomenclature

Variables

A	Cross sectional area
BP	Brake power
c_p	Specific heat for constant pressure
c_v	Specific heat for constant volume
c_w	Specific heat for the matrix
d_m	Wire diameter
f	Friction coefficient
g	Acceleration gravity
h	Coefficient for convective heat transfer
h_{apps}	Pressure head in the connecting application
k	Ratio of swept volume between compression and expansion regions (V_c/V_e)
m	Mass
N	Engine speed
NTU	Number of transfer units
p	Pressure
P_{apps}	Power of the connecting application
p_{max}	Maximum pressure of the working fluid
Q	Heat
Q_{apps}	Flow rate in the connecting application
Q_r	Required heat
Q_{tr}	Real heat transferred through heat exchangers
T	Temperature
U	Velocity of the working gas
V	Swept volume or dead volume for the selected region
W	Work
Δp	Pressure drop
ε	Regenerator effectiveness
η_{apps}	Efficiency of the connecting application
η_{th}	Indicated thermal efficiency
ρ	Density of the working gas
ρ_{apps}	Density of the fluid in the connecting application
ψ	Mesh porosity

Subscripts

c	Compression section
e	Expansion section
h	Heater section
k	Cooler section
r	Regenerator section
w	Wire matrix section

This issue is due to widespread concern for the existing environment and the increase in various types of pollution caused by combustion of fossil fuels and other non-renewable energy sources. One way to achieve this goal is by utilising renewable energy sources such as the sun, wind, tides and rivers to operate vehicles, machines and generators traditionally powered by non-renewable sources, while maintaining clean technology.

Studies about high temperature Stirling engines have been extensively reported in the literature and commercial units have been in operation for many years. On the other hand, low temperature Stirling engines are not as successful as their high temperature counterparts. However, the former have gained popularity in the last few decades due to this potential to tap a variety of low concentration energy sources available, such as solar. The increasing interest in Stirling engines is largely due to the fact the engine is more environmentally friendly than the widely used internal combustion engine, and also to its non-explosive nature in converting energy into mechanical form and thus leading to silent and cleaner operation, which are essential for special applications, such as military operations and medical uses.

Several analyses and simulation methods of the engine have been established [1], as well as the procedures for optimal design [2]. Most of the engines are fuel-fired and operate at high temperature, which highlights the need for careful material selection as well as good cooling system. For silent, light and portable equipment for leisure and domestic uses, low power engines may be more appropriate [3]. Nevertheless, research in Stirling engine technology has been heavily masked by extensive and successful development of internal combustion engines, which have made Stirling engines less competitive [4]. Hence, in order to design a low power engine using solar, new design specifications and optimisation criteria must be established [5]. This paper presents design considerations which may be taken to develop a double-acting Stirling engine operating under a low temperature difference of 50 °C.

2. Mathematical background

Based on the thermodynamic processes, there are three stages of analysis for Stirling engines, namely, the first order isothermal Schmidt analysis [6]; the second order adiabatic analysis [7]; and, the third order analysis [8]. This paper uses the first and the third analyses in developing the engine.

The classical Schmidt analysis was one of the most widely used, and yet the simplest, method in designing Stirling engines. However, it is subject to certain limitations, such as the working gas is assumed to be under the isothermal condition at any instance and location. This assumption leads the internal void volume of the regenerator to be implicitly taken as zero and the successive expansion and compression processes to be always isothermal. Nevertheless, this analysis is still capable of giving good estimation for several critical engine parameters, such as bore dimension, power required and rotation frequency, during the preliminary design stage [5].

After obtaining the estimated values of the engine parameters, the third order analysis is used to study the effect of the pressure drop in the heat exchangers, which include both the cooler and the heater as well as the regenerator, and to calculate the optimal sizing of

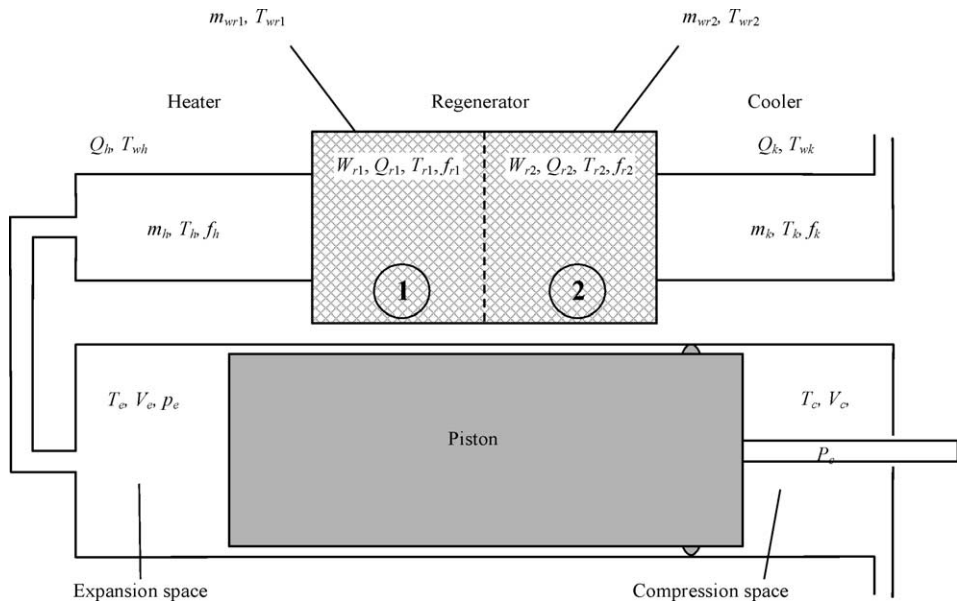


Fig. 1. Schematic model used for thermodynamic analysis.

the engine components including the cylinder bores and all the heat exchangers. The schematic model used in the thermodynamic analysis of the system is shown in Fig. 1. In this figure, the symbols m , T , W , Q and p represent mass, temperature, work, heat and pressure, respectively, whereas the subscripts e , c , h , k , r and w represent sections for expansion, compression, heater, cooler, regenerator and wire matrix, respectively. The assumptions made in the analysis are as follows.

1. The instantaneous pressure is uniform throughout the engine system and the pressure drop in the regenerator is evaluated separately from the thermodynamic analysis.
2. The temperature distributions of the working gas and the matrix in the regenerator are linear, and the regenerator is perfectly insulated.
3. The working gas satisfies the properties of a perfect gas with low velocity.
4. Heat transfer and flow friction in the heat exchangers, i.e. the heater, the cooler and the regenerator, are evaluated using empirical equations under steady flow condition.
5. No leakage is allowed either through the appendix gap or through the seals of the connecting rods.

The heat transfer and the friction losses of the flow of the working gas in the engine components are calculated by using the mass and energy conservation equations. The generalised energy equations for all regions shown in Fig. 1 can be summarised as follows:

$$c_v \frac{d}{dt} (m_e T_{e1}) = c_p T_{eh} \dot{m}_{eh} - A_e h_e (T_{we} - T_e) - p \frac{dV_e}{dt} - \frac{dQ}{dt} \Big|_{\text{losses}_1} \quad (1)$$

$$c_v \frac{d}{dt} (m_h T_h) = c_p T_c \dot{m}_{eh} - c_p T_{hr1} \dot{m}_{hr1} + A_h h_h (T_{wr1} - T_h) - \frac{dQ}{dt} |_{\text{losses}_2} \quad (2)$$

$$c_v \frac{d}{dt} (m_r T_r) = c_p T_{hr2} \dot{m}_{hr2} - c_p T_{r1r2} \dot{m}_{r1r2} + A_{r2} h_{r2} (T_{wr2} - T_{r2}) \quad (3)$$

$$c_v \frac{d}{dt} (m_{r2} T_{r2}) = c_p T_{hr2} \dot{m}_{hr2} - c_p T_{r1r2} \dot{m}_{r1r2} + A_{r2} h_{r2} (T_{wr2} - T_{r2}) \quad (4)$$

$$c_v \frac{d}{dt} (m_l T_l) = c_p T_{r2l} \dot{m}_{r2l} - c_p T_{r2l} \dot{m}_{r2l} + A_l h_l (T_{wl} - T_l) - \frac{dQ}{dt} |_{\text{losses}_2} + \frac{dQ}{dt} |_{\text{losses}_1} \quad (5)$$

$$c_v \frac{d}{dt} (m_c T_c) = c_p T_{lc} \dot{m}_{lc} - A_c h_c (T_{wc} - T_c) - p \frac{dV_c}{dt} \quad (6)$$

where c_v and c_p are specific heats for constant volume and constant pressure, respectively, while Q_{losses_1} , Q_{losses_2} , A and h represent the heat loss from the expansion space to the cooler, the heat loss at the regenerator housing and the matrix due to conduction as evaluated in [9,10], the cross sectional area and the coefficient for convective heat transfer, respectively. In addition to the energy equations, the equations which govern flow continuity across all engine components in Fig. 1 are:

$$\frac{dm_e}{dt} = -\dot{m}_{eh} \quad \frac{dm_h}{dt} = \dot{m}_{eh} - \dot{m}_{hr1} \quad (7)$$

$$\frac{dm_{r1}}{dt} = \dot{m}_{hr1} - \dot{m}_{r1r2} \quad \frac{dm_{r2}}{dt} = \dot{m}_{r1r2} - \dot{m}_{r2l} \quad (8)$$

$$\frac{dm_l}{dt} = \dot{m}_{r2l} - \dot{m}_{lc} \quad \frac{dm_c}{dt} = -\dot{m}_{lc} \quad (9)$$

Across the regenerator matrix, the energy conservation principle is governed by the following equations:

$$c_w m_{wr1} \frac{dT_{wr1}}{dt} = A_{r1} h_{r1} (T_{r1} - T_{wr1}) \quad (10)$$

$$c_w m_{wr2} \frac{dT_{wr2}}{dt} = A_{r2} h_{r2} (T_{r2} - T_{wr2}) \quad (11)$$

where c_w is the specific heat for the matrix. By denoting V as the dead volume for the selected region, the temperature in the regenerator can be derived using these equations:

$$T_{r1} = \frac{2(T_{r1} - T_{r2})V_{r1}}{V_{r1} + V_{r2}} \ln \left[\frac{(2V_{r1} + V_{r2})T_{r1} - V_{r1}T_{r2}}{V_{r2}T_{r1} + V_{r1}T_{r2}} \right], \quad (12)$$

$$T_{r2} = \frac{2(T_{r1} - T_{r2})V_{r2}}{V_{r1} + V_{r2}} \ln \left[\frac{V_{r2}T_{r1} + V_{r1}T_{r2}}{(V_{r1} + 2V_{r2})T_{r2} - V_{r2}T_{r1}} \right]$$

Up to this stage, there any effect from pressure drop as a result of flow resistance by the regenerator matrix is not included. The corresponding pressure drop can be calculated as follows:

$$\Delta p = p_e - p_c = \frac{1}{2}f_h\rho_h U_h^2 + \frac{1}{2}f_{r_1}\rho_{r_1} U_{r_1}^2 + \frac{1}{2}f_{r_2}\rho_{r_2} U_{r_2}^2 + \frac{1}{2}f_k\rho_k U_k^2 \quad (13)$$

where f , ρ and U represent friction coefficient, density and velocity for the working gas in each particular region, respectively. Consequently, the indicated thermal efficiency η_{th} can be written in terms of the ideal adiabatic efficiency as follows:

$$\eta_{th} = \eta_{ideal} - \eta_{ideal} \int \Delta p \, dV \quad (14)$$

3. Design specification and concept

3.1. Engine specification

The engine parameters should be optimised to avoid losses and to obtain high thermal efficiency for all the engine components especially heat exchangers. While the main target of the engine is to produce sufficient power to run a connecting application, there are conditions which pose critical constraints on the design, i.e. the working fluid is air and the temperature difference between the heater and the cooler is about 50 °C only. It is desirable that the engine is able to run continuously without assistance from any secondary power source, thus a Siemens configuration is chosen, which includes four double-acting cylinders and pistons, heaters, coolers and regenerators. All four units of cylinder and piston, supported by a top shaft are allowed to swing laterally. This swinging mechanism introduces a slight deviation from the conventional slider-crank mechanism whereby, similar to the function of energy wheels, the mechanism provides additional inertia, which may be needed to maintain the rotating motion. Another deviation, compared the conventional configuration, is that the connecting rod is fixed to the piston and the reciprocating action of the four pistons in certain sequence transfers the motion to the crankshaft which can be connected to energy wheels via flexible belts. The main engine specifications are listed in Table 1.

3.2. Design concept

In order to estimate the design parameter for engine performances, the first step which needs to be taken is to find the required power to operate the target application, which is in

Table 1
Specifications and target performance

Parameters	Values/Type
Engine type	Alpha-Siemens
Working fluid	Air at STP
Engine speed	100–130 rpm
Output power	100 W

the form of its brake power (BP) of a connecting application. In the case of a water pump, the brake power can be estimated by

$$BP = \frac{P_{\text{apps}}}{\eta_{\text{apps}}} = \rho_{\text{apps}} g h_{\text{apps}} Q_{\text{apps}} \quad (15)$$

where g is acceleration gravity (9.81 m/s^2), ρ_{apps} is density of the water (1000 kg/m^3), h_{apps} is the design head of the pump (10 m), Q_{apps} is the flow rate ($2.40 \text{ m}^3/\text{hr}$), η_{apps} is the pump efficiency (80%) and P_{apps} is the power of the water pump. From the above Eq. (15), the approximate brake power required to run the selected application is 82 W. In order to achieve this value, the system should be designed so that any losses in heat exchangers, regenerators, driving mechanism, seals and appendix gaps are minimised. Like any other Stirling engine, the efficiency may approach a value of slightly more than 50% that of the thermodynamically ideal Carnot value [11]. Thus, the indicated power, which represents power to be produced inside the cylinders without any losses based on the Schmidt analysis, should be at least twice that of the brake power. As a result, the wire-frame drawing representing the design concept of the Stirling engine is developed as shown in Fig. 2 where the internal configuration of the engine is given in Fig. 3.

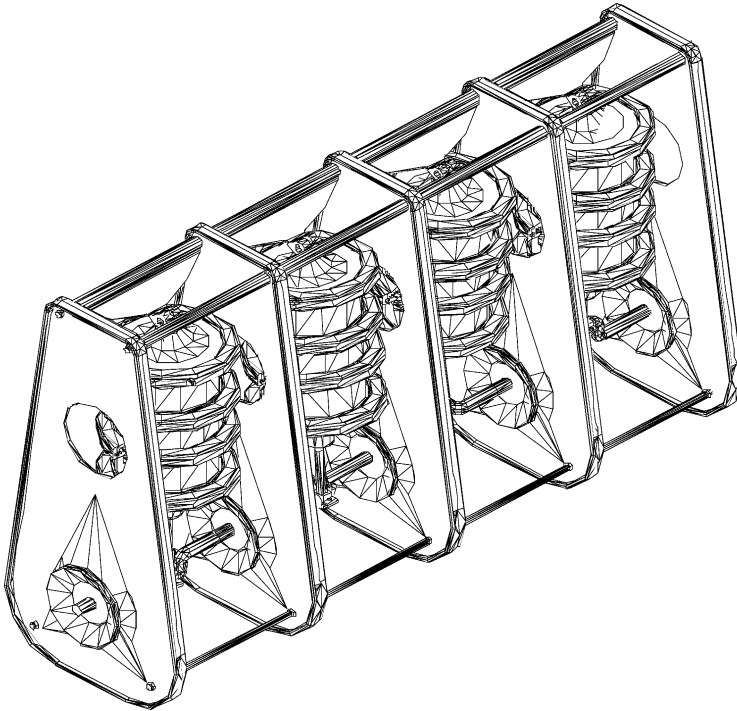


Fig. 2. Wire-frame model for the conceptual design of a Stirling engine.

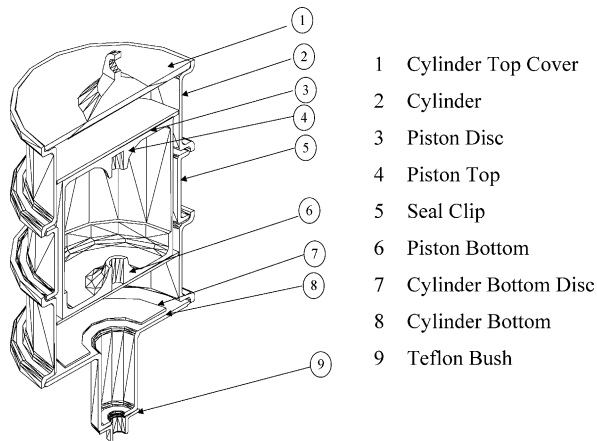


Fig. 3. Internal components of the cylinder.

3.3. Material selection and processing

The main challenge during the design process is that the power is low compared to the engine size, which may be relatively large. Hence, material selection plays an important role in order to make the engine operating at an optimal level while the structure is strong enough to support the whole structure and the entire engine is cost competitive. All factors, such as strength, friction, thermal consideration, cost, safety, shape, weight, stiffness, surface, finish and lubrication, should be taken into account. In this work, most of the parts and components are made of glass fibre reinforced polyester (GFRP), which is cheaper and easier to mould than metal. Certain precautions must be taken for critical components which have thickness variation, sharp curvatures, several number of cavities and reinforcements layers [12,13]. However, GFRP is not suitable for use in some of the engine parts, such as the crankshaft and the connecting rods, since these parts are subjected to bending and torsional loading [14], they are made of aluminium. As these are the dominant stress modes, equations used for component design should adopt a safety factor of at least 1.5. The length of connecting rods should be calculated to avoid buckling. After carrying out the stress analysis according to [15], the wall thickness of the cylinder should be at least 6 mm, and the fillet radius between 3 and 8 mm for all the components made from GFRP.

During fabrication, since the GFRP material is considered of high density, all parts made from GFRP should be of hollow design in order to reduce weight and cost. The piston is designed with top and bottom parts to enable hollow sections formed inside the piston. The top and bottom surfaces of the piston are covered by a type of flexible seal which is fixed onto the piston surfaces via light disc. This seal is introduced to avoid any leakage of hot or cool air through the appendix gap. In order to avoid any vibration to the piston and to reduce the friction between the cylinder and the piston, the top edge of the light disc contains miniature grooves for nylon o-ring. This will reduce friction and also air leakages through the appendix gap. The bottom part of the pistons is connected to

the connecting rod and eventually to the crankshaft. The whole configuration of the cylinder and the piston is allowed to swing with respect to the top supporting brackets. Mass eccentricity is not an issue due to its low operating speed. The crankshaft is supported by nylon bushes positioned at side flanges also designed from the GFRP, where these nylon bushes are used to reduce friction between the crankshaft and the flanges.

4. Design analyses

The analyses can be subdivided into four stages: the analysis of swept volumes and dead volumes; the analysis of heater and cooler parameters; the analysis of regenerators; and, finally, the optimisation the critical engine parameters.

4.1. Relationship for engine power, swept volumes and dead volumes

The purpose of this simulation is to estimate the main volumes of the engine spaces in terms of swept volumes and dead volumes using the Schmidt analysis since these factors are essential in estimating the preliminary configuration of the engine and will influence the subsequent optimisation process. Since it has been decided to adopt the successive alpha-type four-cylinder Siemens configuration, the compression swept volume V_c should be equal to the expansion swept volume V_e , and thus the swept volume ratio $k = V_c/V_e = 1$. In addition, at this stage, it is assumed that the mean pressure of the engine during operation is the standard atmospheric pressure of 101.3 kPa (abs), which is the kind of pressure which normally occurs before the engine start-up. In addition, the phase angle of the expansion space varies relative to the compression space and the phase angle should be 90° and is constant throughout the calculations.

It is obvious from Schmidt-cycle equations that the net cycle power and the thermal load on the heat exchangers are direct linear functions of the engine speed (N), the maximum pressure of the working fluid ($p_{\max}$) and the size of the engine, which is expressed in term of the swept volume [5,7]. However, the direct effects of the dead volume and swept volume to the engine power should be detailed. Fig. 4 illustrates the variation of the power as a function of the swept volume, which was calculated on the dead volumes of 2.5 and 5.0 l under the fixed temperature difference of 50°C . It is shown that the power increases when the swept volume increases. Also, it is noticeable that the speed increases with the increase in power. These two remarks imply that the swept volume is proportional to the engine power for several speeds. By comparing the two graphs in Fig. 4, based on the same swept volume, it can be said that the decrease in dead volume will lead to an increase in engine power.

To illustrate the effect of the dead volume clearly, the variation of the engine power as a function of dead volume is calculated and the results are as shown in Fig. 5 for the engine speed of 120 and 80 rpm.

From Figs. 4 and 5, it can be seen that the increase in the dead volume produces an exponential drop in the net power, which in turn decreases the maximum pressure. However, the calculation is performed under the assumption that the temperature difference is 50°C , which can be obtained from the solar collector and the water tank.

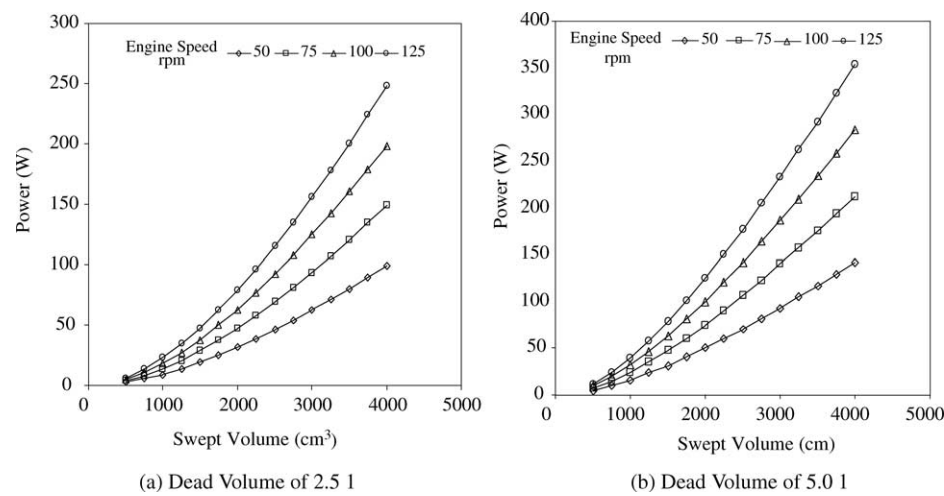


Fig. 4. Relationship between swept volume and engine power.

4.2. Relationship for heater and cooler parameters

An important factor in heat exchanger design is volume. Cooler and heater volumes contribute to large portions of dead volume. Previous studies showed that the dead volumes, which includes those in the heat exchangers, is an essential factor in the Stirling engine design, where it should be small as possible [16,17]. To demonstrate the relationships for the heaters, specific conditions of 3.2 l swept volume and 0.25 m tube length are used.

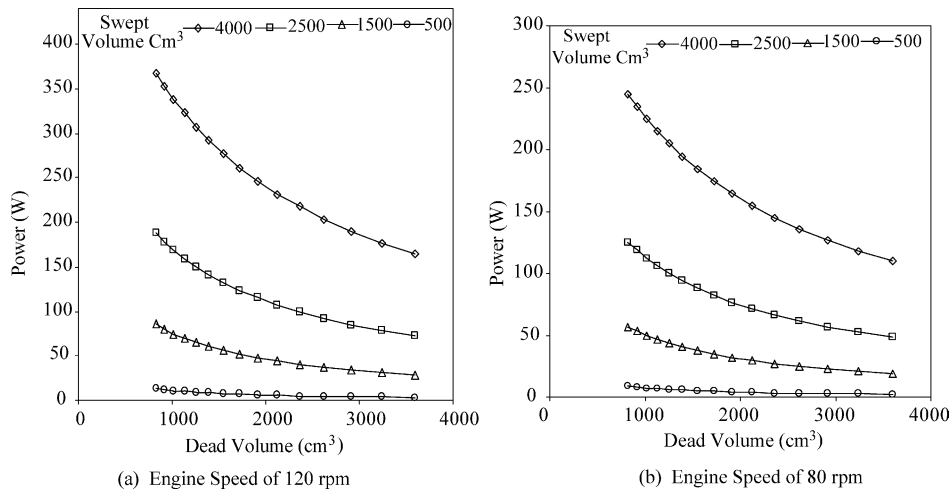


Fig. 5. Relationship between dead volume and engine power.

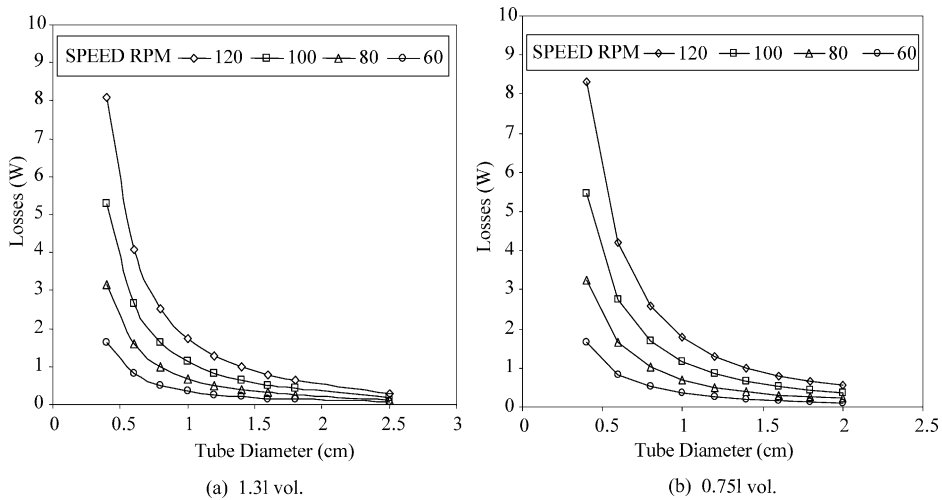


Fig. 6. Relationship of heater tube diameters with the friction losses. (Swept volume=3.2 l, tube length=0.25 m).

After carrying out thermodynamic simulation for the heater, the variation of its tube diameter can be derived as a function of friction losses for several values of engine speeds as being depicted in Fig. 6(a) for the heater volume of 1.3 l. Similarly, Fig. 6(b) shows the graphs for the heater volume of 0.75 l. Both graphs indicate an inverse proportionality between tube diameter and friction loss in the heater. The explanation of this variation is that the tube with smaller diameter having the same length delivers the same mass flux, thus generates a shorter entrance length and a thicker viscous boundary layer, which then leads to a higher friction factor of the flow. For the cooler, by using the same values for swept volume and tube length, the equivalent graphs for the cooler volumes of 1.3 and 0.75 l are shown in Fig. 7, which indicates a similar pattern to that of the heater.

In designing heat exchangers, an important consideration for the heat exchangers is to have an ability to supply or reject the required amount of heat to or from the engine. In this aspect, one crucial factor is the heat transfer area, which will decide the amount of heat energy to be transported. Hence, in order to achieve a high effectiveness for the heater and the cooler, larger transfer areas, and thus larger volumes, are needed. After carrying out the analysis on the heat flow through the heat exchangers, the relationship between the ratio between the required heat Q_r and the real heat Q_{tr} transferred through the heat exchangers with respect to the tube diameter for the two heater volumes analysed previously are shown in Fig. 8. This analysis shows that a smaller tube diameter is needed to achieve the ratio of Q_r/Q_{tr} close to unity and an increase in the heater volume leads to a better ratio. This relationship is opposite to the general pattern shown in Figs. 6 and 7. Hence, a suitable value for tube diameter for an optimal design and in this case, a tube diameter of 15 mm and the heater volume of 1.3 l, has been selected. If the similar analysis is repeated for the cooler, the same pattern and results are obtained.

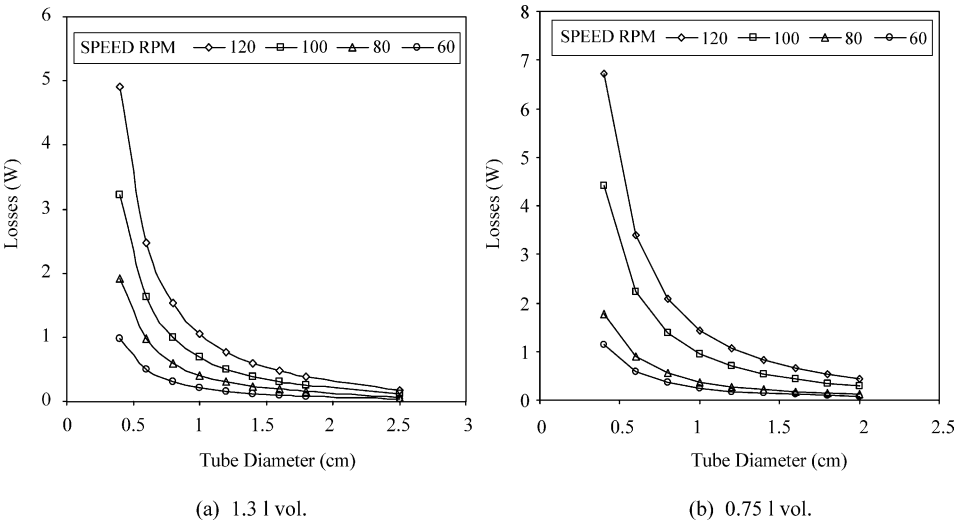


Fig. 7. Relationship of cooler tube diameters with the friction losses. (Swept volume = 3.2 l, tube length = 0.25 m).

4.3. Relationship for regenerator parameters

The effect of pressure drop in the regenerator of a low temperature differential Stirling engine to thermal efficiency is very important since it can decrease the overall efficiency of

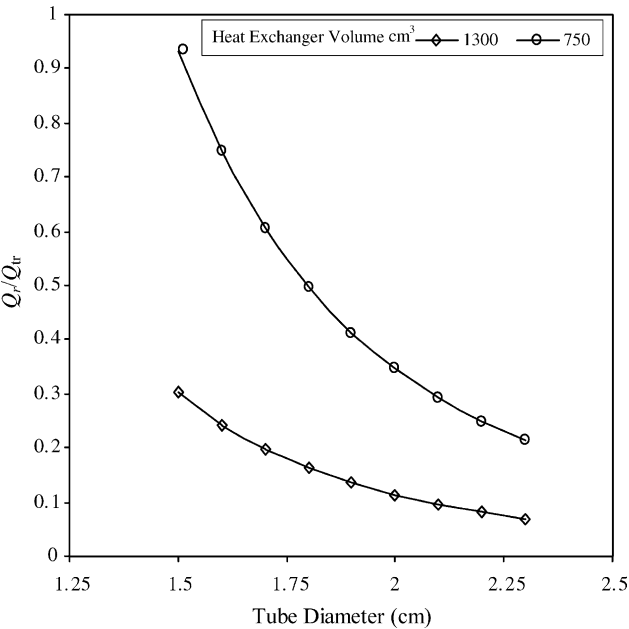


Fig. 8. Relationship between ratio of required heat and tube diameter.

Table 2
Geometrical properties of wire mesh for regenerator

Matrix	Wire diameter d_m (m)	Porosity ψ
M1	0.0035	0.9122
M2	0.0050	0.8359
M3	0.0065	0.7508
M4	0.0070	0.7221
M5	0.0080	0.6655
M6	0.0090	0.6112

the engine [16,18]. To analyse this effect and its implication to the efficiency of the engine, six types of matrices has been selected and is being subjected to various pressure drops and engine speeds. The configurations for these six matrices are given in Table 2 for a standard total wire length of 5 m. The porosity of each matrix is important since it will have a direct impact on the performance of the regenerator, and can be determined by its geometry, namely, wire diameter, density of the mesh and the void volume. Any changes in the porosity will also change the regenerator effectiveness and the pressure drop, which eventually affects the engine efficiency. Therefore, the best matrix for the regenerator should possess both high effectiveness and low-pressure drop.

Fig. 9 shows the relationship between the engine speed and the pressure drop for these matrices. The pressure drop is found to be proportional to the engine speed since an

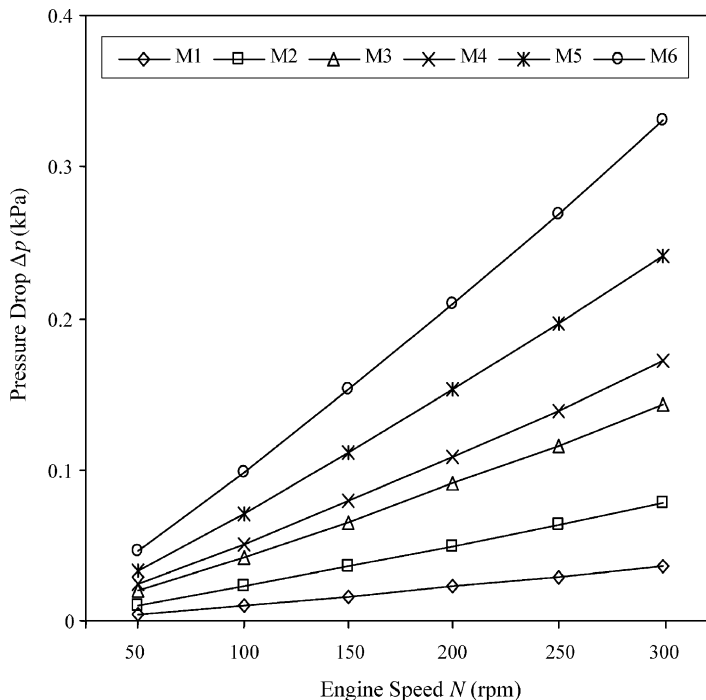


Fig. 9. Relationship between pressure drop and engine speed.

increase in engine speed increases the mass flux through the regenerator as well as the pressure magnitude up to the same proportion for the same matrices. On the other hand, the decrease in mesh porosity leads to the higher friction factor as well as increases the pressure drop. Hence, it can be said that M1 has a lowest pressure drop in comparison to the others at a same speed because its porosity is the highest. In order to obtain a higher porosity, and thus the lower pressure drop, the meshes should be made from small wire diameter and should be as coarse as possible. However, the pressure drop in the regenerator alone is not sufficient in deciding the best regenerator without considering its heat transfer behaviour.

The regenerator effectiveness ε can be manipulated by varying wire diameter and wire length, which in turn changes the ‘wetted’ surface area. It can be represented in form of the relationship between the porosity or the number of transfer units (NTU) and the thermal heating efficiency of the engine. If the wetted surface area is large, the resulting porosity should be low, and this provides the air or the work fluid with a large contacting surface to achieve a high rate of heat transfer. Hence, the NTU, and thus ε , are increased when the surface area increases. The effect of ε on the thermal efficiency of the engine is that it represents the ability to reject the heat to the working gas when the gas exits through the heater and the ability to absorb the heat when the gas exits through the cooler. Therefore, the next step is to identify the configuration for the best mesh having low pressure drops but at the same time possessing high effectiveness.

Fig. 10 shows the relationship between the regenerator effectiveness with respect to the engine speed, where it appears that ε in M1 is lower than that for M6 for all engine speeds. This implies that the effectiveness of the regenerator is better for a coarser mesh than that for a finer mesh. Also, the effectiveness is better for an engine running at low speed rather than at higher speed since heat transfer activities can take place more efficiently in the low speed condition.

The effect of the thermal losses in the regenerator can be addressed by considering the effect of both parameters on the engine thermal efficiency. Fig. 11 shows the relationship between the thermal efficiency and the engine speed, and this parameter be seen as inverse to the percentage of the losses due to the pressure drop and the effectiveness. Therefore, the best matrix should compromise between high effectiveness and low-pressure drop in order to obtain minimal losses in the regenerator, and in this case, M4 with the porosity of 0.722 has been chosen for the design.

4.4. Relationship for engine power and cylinder bore

In this section, the optimisation of critical engine parameters, using the relationship established in the previous sections, other parameters can be determined accordingly based on these optimal parameters. In a typical Stirling engine design, swept volume forms a linear function with engine power as established in the ideal Schmidt analysis and the third order analysis [7,8]. Furthermore, from the Schmidt analysis, the dead volume should be as small as possible to obtain a higher power. However, the Schmidt analysis neglects losses attributed from pressure drops and heat transfers. By using the third order analysis [8], issues regarding the effects from pressure drop and heat transfers to the engine efficiency can be addressed.

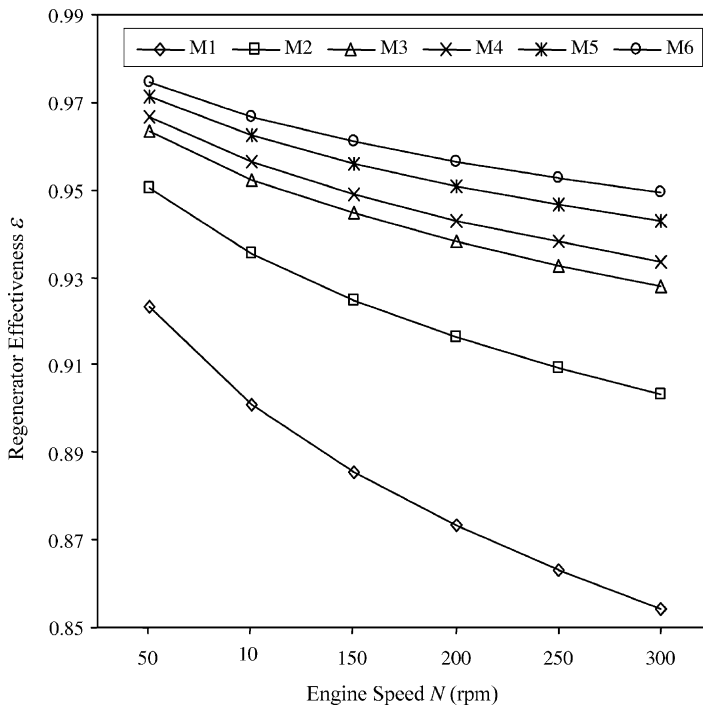


Fig. 10. Relationship between regenerator effectiveness and engine speed.

On carrying out the analysis, a linear relationship can be clearly seen in Fig. 12 for the $Q_{in}-Q_{out}$ term, which can be regarded as ideal power, with the engine speed since it neglects any losses due to pressure drop or mechanical inefficiency. However, since the losses also increase with the speed as shown in Fig. 12, thus the net power decreases after it reaches a maximum value between 100–150 rpm. This variation can be a good basis for design optimisation.

Based on the optimal value of engine speed identified from Fig. 12, Fig. 13 presents the relationship between engine sizes, which is expressed in term of the bore diameter, with the thermal efficiency for both the ideal Schmidt analysis and the third-order flow analysis. The Schmidt analysis indicated that the thermal efficiency is independent of the bore diameter because it only considers the temperature difference, while third-order analysis does include the effects from pressure drop and heat transfer. All the bore configurations have been analysed to give the same indicated power with an optimal speed 120 rpm.

After calculating the maximum power of the engine to run the system, one must also consider the minimum size of the engine in the design to produce this power. Generally, the temperature difference is a source of the power function, and the maximum temperature difference that can be achieved by this system of 50 °C. From the thermodynamic analysis of the Stirling engine, the design of the heat exchangers and regenerator volumes can be optimised to produce high engine efficiency at low friction and thermal losses.

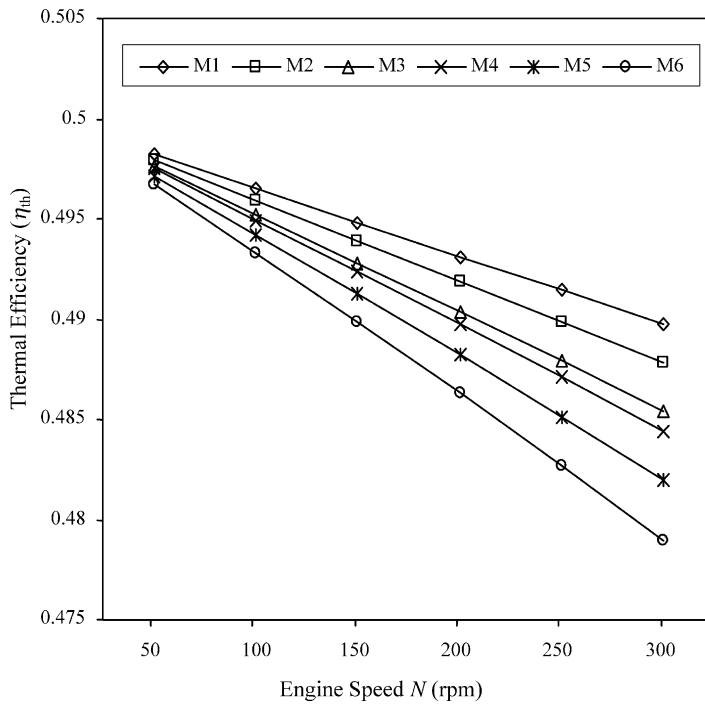


Fig. 11. Relationship between thermal efficiency and engine speed.

5. Parametric optimisation

It is understood that the heat transferred by a smaller size of heater or cooler will not be adequate to run a low temperature differential Stirling engine due to the size of the cylinder bore. On the other hand, the dead volume will introduce deficiency in Stirling engines for both ideal and actual cycles. Even in the ideal Stirling engine, there should be some default void volumes as in the regenerator and other heat exchangers. By including these dead volumes, it reduces the amplitude of pressure excursion when the working gas moves from the cold space to the hot space. Eventually for all these losses, which have been produced in heat exchangers, many values of cylinder bore diameter have been calculated at different dead volumes in order to get an optimal value.

Based on the optimal speed of 120 rpm, the sizing for all heat exchanger components can now be decided. Hence, the optimal parameters for the design are found to be the heater, cooler and regenerator volumes of 1.3 l, 1.3 l and 2.0 l volumes, respectively. For this case, the losses of the heat exchangers and the regenerator are very high. Therefore, an increase in the heat exchangers and regenerator volumes should be accompanied by an increase in the swept volume to produce a required power. From the Schmidt analysis, the minimum swept volume should be 1.0 l at the reference dead volume of 2.5 l. In these conditions, the optimal swept volume should be 3.2 l. Therefore, the bore diameter of 0.20 m has been selected for the design. Also, during the analysis for heat exchangers,

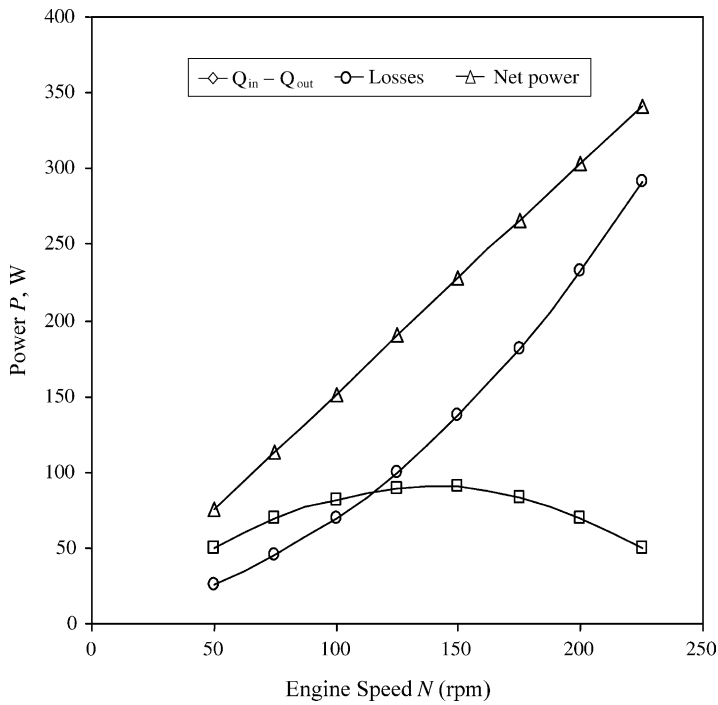


Fig. 12. Relationship between engine speed and overall power.

the losses are found to be relatively steady at the bore diameter of 0.20 m. The details of the heat exchanger configurations are shown in Table 3.

In typical engine design, the distance of the stroke is usually designed to be two times of the bore diameter of the piston [19], and thus the dimensions of the stroke are made to be 0.10 m. Consequently, from stress calculation, the permissible radius of the crankshaft rod is found to be 0.05 m. In terms of internal working pressure, from the previous analysis, the maximum pressure in side the cylinder is found to be 180 kPa (abs) or 79 kPa (gage). Similarly, and the minimum pressure is found to be is 70 kPa (abs) or -31 kPa (suction).

6. Conclusion

In this paper, a number of technical considerations in designing a low temperature differential Stirling engine have been proposed. These considerations have been established through the use of the Schmidt analysis and the third order analysis, and have later been optimised. As a result, the optimal configuration for the design can be summarised as follow.

- Upon optimisation, engine speed has to be limited between 100 and 150 rpm at a temperature difference of 50°C , where the optimal value is 120 rpm.

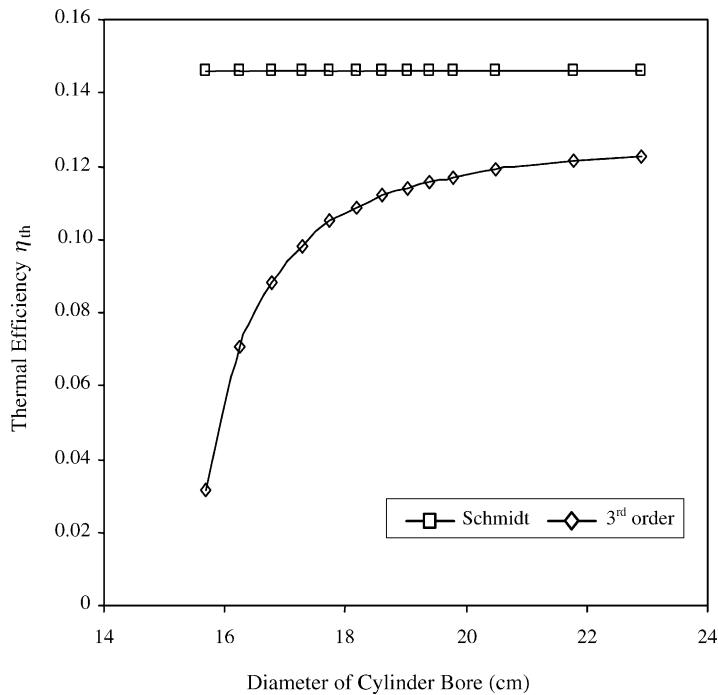


Fig. 13. Relationship between engine efficiency and the cylinder bore.

- For the regenerator, its porosity plays a significant role in controlling its efficiency of the regenerator, which can be manipulated by varying wire diameter and length. For this engine, the selected parameters are the wire diameter of 0.7 mm with a total length of 5 m and a porosity of 0.722 or 72.2%.
- The heat exchanger volumes should be evaluated by considering both the pressure drop and the heat exchanger effectiveness. The optimal heat exchanger volume has been found to be 1.3 l for both the cooler and the heater and 2.0 l for the regenerator, and the tube dimension is 0.015 m in diameter and 0.250 m in length.
- The swept volume should be as large as possible since there is a proportional relationship between the indicated power and the swept volume. In this paper, the optimal swept volume is 3.2 l, which leads to 0.20 m bore piston diameter.

Table 3
Optimal configurations for heaters and coolers

Volume	1.3 l
Tube diameter	15 mm
Tube length	0.25 m
Number of tubes	34
Material	Copper

- After identifying all critical engine design parameters, the engine components such as cylinders, pistons, regenerator compartments and some structural supporters are to be manufactured mainly using GFRP, except for the crankshaft and the connecting rods which are to be manufactured from aluminum alloy.

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